Amendments to the Specification:

The paragraph starting at page 2, line 14, is amended and now reads as follows:

-- The essentially non-linear material characteristics of the rubber can be disadvantageous in this context. For example, the stiffness of a rubber element increases with larger deflections and is essentially caused by its significant transverse expansion. As a consequence, the resonance frequency of the vibrating system made up of the motor unit, the handle, the intermediately connected antivibration element or device changes in dependence [[of]] upon the preload and the vibration amplitude. An adaptation of the resonance frequency to the operating frequency range of the work apparatus is therefore difficult. An operation of an antivibration element of this kind in a quasi-linear range is only possible for a correspondingly large configuration of the antivibration element for which sufficient mounting space is not always available. High operating loads or tight spatial conditions require the arrangement of a vibration damper, for example, in a sleeve, which prevents the transverse expansion of the damper material. Blocking the transverse expansion leads, with rubber, to a considerable stiffening, which makes an adaptation to the excitation frequencies to be dampened difficult. --

The paragraph starting at page 4, line 29, is amended and now reads as follows:

-- In the loaded state, the mass of the vehicle body and the static loading of the isolation elements [[is]] are increased. The stiffness of the PU-foam increases. In combination with the increased mass of the vehicle body, the natural frequency can at least be held approximately constant independently of the static loading state. The natural frequency of the vibration system can be determined under static considerations. However, the PU-foam damper performs in this range like a conventional rubber damper whose stiffness likewise increases with the static load. With respect to the dynamic matching, no special advantages can be achieved with the use of a PU-foam damper compared to a rubber damper. --

The paragraph starting at page 16, line 24, is amended and now reads as follows:

-- The motor unit 1 and the tubular handle 11 with the intermediately disposed antivibration elements 5 of FIG. 1 form a vibrating system whose vibration excitation takes place essentially because of the engine 2 and the saw chain (not shown). The vibration response in the tubular handle 11 dependent upon the excitation frequency is shown, by way of example, in the form of a diagram in FIG. 5. The trace of the vibration response is shown as an enlargement function $V_D$ referred to the 100% line of the excitation amplitude. The useable rpm range of the engine 2 of FIG. 1 lies between the idle rpm of approximately 3,000 rpm and the full-load rpm of approximately 12,000 rpm. This corresponds to an excitation

frequency range $f_B$ from 50 to 200 Hz to be damped. The excitation frequency range of the saw chain lies, in the embodiment shown, in the range between 50 and 80 Hz and therefore within the ~~frequence~~ frequency range $f_B$ which is generated by the engine 2. --